Laid-Open Publication

DE 1 576 255

Reciprocating and Rotary Piston Internal Combustion Engines with a High Level of Efficiency

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Reciprocating and rotary piston internal combustion engines are built as Otto engines and as Diesel engines. As Otto engines they function with spark ignition and a compression ratio of approximately 7.5 to 11:1. Diesel engines have compression ignition and have a compression ratio of approximately 20:1. The maximum achievable efficiency level for Otto engines is approximately 30 %, and for Diesel engines approximately 50 %. The losses are made up of heat emission and engine cooling losses, exhaust losses and friction and control losses. They are caused by the functional [or operational] mode of the two systems, and with the same functional mode, they can only be reduced by increasing the compression ratio because the greater compression with the correspondingly higher compression heat brings about a steeper drop in temperature between the start and the end of the expansion stroke which determines the level of efficiency. There are limits, however, to the increase in compression, and with the Otto engine due to the compression ignition of the petrol [or gasoline]/air mixture at high compression heat, and with the Diesel engine due to the very high compressive load of the material with a compression ratio of over 20:1. In order to achieve good performance and efficiency levels, Otto engines with compression greater than approximately 8:1 must therefore already use the fuel with a particularly high octane rating known as “super petrol” [or "premium gasoline"] in order to avoid compression ignition.

With this prior art, the efficiency level of the Otto and Diesel engines still using the traditional working methods can only be unsubstantially changed. For years, therefore, the engine output per litre has increased, but not in the same proportion as its level of efficiency. The invention makes it possible to substantially increase the levels of efficiency achievable according to today's prior art, and so to produce more economical engines with lower fuel consumption. It is to be used with reciprocating and rotary piston engines. The invention relates to the control of the air and gas mixture inlet and the combustion gas outlet while at the same time increasing the compression ratio. All

previous reciprocating and rotary piston internal combustion engines are designed such that the cylinders wherever possible are charged to 100 %, and with high performance engines using auxiliary means, and even over-charging. The disadvantages are bad utilization of the gas expansion, high exhaust gas losses due to high gas temperature with corresponding gas pressure at the time of the gas discharge when the expansion stroke ends, high engine cooling losses, and low engine efficiency. With the invention, by changing the inlet control timings, the cylinders are only charged to half their piston displacement or chamber volume with combustion air or fuel/air mixture. At the same time, the compression ratio with respect to the normal engine design is increased to double so that, in relation to the piston displacement, it would be approximately 15 to 22:1 with the Otto engine, and approximately 40:1 with the Diesel engine. Because, however, the cylinders are only half charged, this corresponds in reality to the normal compression ratios of 7.5 to 11:1 with Otto engines, and 20:1 with Diesel engines. The expansion stroke following the compression of the half charge does not just extend over half the stroke path (corresponding to the half cylinder charge), but over the complete stroke path from top to bottom dead center. In this way, it is possible for the first time to extend the final expansion of the compressed combustion gases of an engine running with a full load before the expansion stroke to the possible limit by approximately 100 %, and in this way to better utilize the expansion of the combustion gases than with the previous working methods. By means of the additional expansion, the combustion gases are better relieved, and the pressure otherwise present at the start of the gas discharge and the high waste gas temperature are substantially reduced. The average gas temperature between intake and discharge of the gases is lower than with conventional functional modes due to the steeper temperature drop during the expansion stroke, and so the engine must not be so strongly cooled. Because the level of efficiency is determined by the drop in temperature and pressure caused by the work output, a higher level of efficiency must inevitably be achieved by the invention with its effects upon the drop in temperature and pressure. The increase in performance results from the gas expansion in the second stroke half of the expansion stroke (the first stroke half corresponding to the half cylinder charge), from the lower waste gas

losses as a result of lower gas pressure and lower gas temperature at the time of gas discharge, and from the smaller engine cooling losses. The fuel consumption drops due to the half charge, but the engine performance is higher than would correspond to the half charge. The waste gases have fewer non-combusted components due to the comparatively longer combustion and expansion. In addition to this, there is also the possibility of undertaking the combustion with excess air, without fear of the engine over-heating because, as mentioned above, the average gas temperature is lower than with previous engines. Combustion with excess air does, however, go hand in hand with an improvement in combustion. As a result, this means for the invention not only achieving a high level of efficiency, but also providing a method for keeping the air clean. The installation of several devices, which only serve the purpose of keeping the air clean, becomes unnecessary if, by the use of the invention, correspondingly good combustion is guaranteed.

In order to achieve the half charge, there are basically two possibilities. In one case, the gas is sucked in over the whole piston stroke, and during the compression stroke the excessive half charge sucked in is pushed back out of the cylinder. In the other case, the gas is only sucked in over the half piston stroke.

Implementation of the invention with the different types of engine:

1.) <u>Valve-controlled four stroke engines</u>

a) The compression chamber is reduced to 50% of its normal size so that despite the half cylinder charge, the gas sucked in is compressed to the desired, normal ratio. The inlet valve closes after approximately 270° crank shaft rotation when the piston, after the full induction stroke, has already implemented the half of the compression stroke with the inlet valve still open, and the excessive gas sucked in for the half charge is pushed back out of the cylinder into the induction pipe. The half charge which remains in the cylinder is compressed to the normal ratio during the second half of the compression stroke with the help of the reduced compression chamber. In the following

expansion stroke, the gas has the possibility of extended expansion because the outlet valve only opens at bottom dead center (after 180° crank shaft rotation), and the gas can expand to double its induction volume. The gas pushed back into the induction line is taken by the other cylinder which is just sucking in.

b) The inlet valve already closes after approximately 100° crank shaft rotation of the induction stroke when the cylinder is half charged. The compression to the normal ratio and the extended expansion correspond to that stated under a).

The advantage of the method described under a) is that the desired cylinder charge is also achieved in the highest range of revolutions.

2). <u>Valve-free four stroke engines (Wankel engines)</u>

The principle of the half charge, of compression to the normal compression ratio despite the half charge, and of the extended gas expansion corresponds to that stated under 1.). The half charge is achieved in that:

a) by means of an overflow duct between the compression chamber and the induction chamber, the excessive charge sucked in from the compression chamber is pushed back into the induction chamber until only the half charge now remains in the compression chamber, and the rotating piston covers the overflow duct so as to prevent further overflow,

b) by means of an exactly adapted form and positioning of the gas inlet opening in the induction chamber, only the half charge can be sucked in until the rotating piston closes the induction opening. In this case, the overflow duct described under a) is dispensed with.

The use of the overflow duct offers a better guarantee of correct chamber charging, even in the highest range of revolutions.

3.) Two stroke engines

In order to achieve the half charge and the extended gas expansion, additional gas control is required so that the overflow duct for fresh gas, which must lie approximately half way between top and bottom dead center, remains closed during the full expansion stroke, and only makes gas exchange possible when the compression stroke begins. The outlet duct is controlled such that it opens shortly before bottom dead center, and only closes again after the gas exchange during the compression stroke. In this way, the engine has from 0 - 180° expansion stroke (from top to bottom dead center), from 180 - 270° outlet stroke, with 270° fresh gas inlet, and from 270 - 360° compression stroke.

Patent Claims

1.) A reciprocating piston internal combustion engine with a high level of efficiency, valve-controlled, working by the four-stroke method, characterized by the simultaneous use of the following features:

a) The inlet valve is open from top dead center to approximately 270°crank shaft rotation, and only closes when the piston, after the full induction stroke, has implemented the half compression stroke with the inlet valve still open, and so has pushed an amount of gas back into the induction pipe such that the cylinder is only half charged.

b) The compression chamber is reduced to 50 % of its volume calculated from the cylinder capacity so that theoretically there is a compression ratio of 15 – 22:1 with Otto engines, and approximately 40:1 with Diesel engines, the half cylinder charge however only being compressed in the compression ratio normal for complete cylinder charges of 7.5 – 11: 1 with Otto engines and approximately 20:1 with Diesel engines.

c) The compression of the half charge is implemented in the second half of the compression stroke between 270 - 360° crank shaft rotation.

d) The expansion stroke extends over the complete piston stroke (= 180° crank shaft rotation) and is therefore double the length compared with the compression stroke implemented with the half stroke according to c).

e) The expansion of the combustion gases is extended by 100 % by the expansion of the half cylinder charge to the whole displaced volume of the cylinder. The expansion of the combustion gases to double their induction volume results in a steep drop in temperature, lower exhaust gas temperature, lower exhaust gas pressure, smaller exhaust loss, smaller engine cooling loss and a higher level of efficiency.

2.) The Claim according to 1.), characterized in that the half cylinder charge is achieved by closing the inlet valve after the half induction stroke, i.e. after approximately 100° crank shaft rotation instead of after 270°, according to 1.) a).

3.) A rotary piston internal combustion engine with a high level of efficiency, valve-free, working by the four-stroke method, characterized by the simultaneous use of the following features:

a) An overflow duct connects the compression chamber to the induction chamber. Its opening lying in the compression chamber is closed by the rotary piston when, during the compression stroke, an amount of fresh gas is pushed back out of the compression chamber into the induction chamber such that the compression chamber is now only charged with the half of its full chamber volume.

b) The combustion chamber is reduced to 50 % of its volume calculated from the chamber size, so that theoretically there is a compression ratio of 15 – 22:1 with Otto engines and approximately 40:1 with Diesel engines, the half chamber charge only being compressed, however, in the compression ratio normal for complete chamber charges of 7.5 – 11:1 with Otto engines and approximately 20:1 with Diesel engines.

c) The compression of the half charge is implemented in the second half of the compression stroke.

d) The expansion stroke extends over a complete stroke and so is double the length of the compression stroke implemented in a half stroke according to c).

e) The expansion of the combustion gases is extended by 100 % by the expansion of the half charge to the complete chamber volume. The expansion of the combustion gases to double their induction volume results in a steep drop in temperature, lower exhaust gas temperature, lower exhaust gas pressure, smaller exhaust loss, smaller engine cooling loss and a higher level of efficiency.

4.) The claim according to 3.), characterized in that in order to achieve the half charge, the gas inlet opening lies at that point in the induction chamber where it is closed by the rotary piston during the induction stroke when the half of the chamber volume is charged with fresh gas. The overflow duct is dispensed with in this case.

5.) A two-stroke reciprocating piston internal combustion engine with a high level of efficiency, characterized by the simultaneous use of the following features:

a) A gas control influences the opening times of the gas outlet duct and the fresh gas overflow duct.

b) Due to its form and position in the cylinder wall, the gas outlet duct is open from approximately 90° - 270° crank shaft rotation. It is influenced, however, by the gas control such that the gas discharge can only happen as from approximately 180° at bottom dead center.

c) Due to its position in the cylinder wall, the fresh gas overflow duct is open both at approximately 90° and shortly before 270° crank shaft rotation. However, the additional gas control only allows fresh gas inlet with at the same time discharge of the residual combustion gases at 270°. The cylinder is only charged with half the displaced volume by the gas exchange at 270°.

d) The compression chamber is reduced to 50 % of its volume calculated from the cylinder capacity so that theoretically there is a compression ratio of 15 – 22:1 with Otto engines and approximately 40:1 with Diesel engines, the half cylinder charge however only being compressed in the compression ratio normal for complete cylinder charges of 7.5 – 11:1 with Otto engines and approximately 20:1 with Diesel engines.

e) The compression of the half charge is implemented in the half stroke between 270 - 360° crank shaft rotation.

f) The expansion stroke extends over the whole piston stroke from 0 - 180° crank shaft rotation and is therefore double the length compared to the compression stroke implemented with the half stroke.

g) The expansion of the combustion gases is extended by 100 % by the expansion of the half cylinder charge to the whole displaced volume of the cylinder. The expansion of the combustion gases to double their induction volume results in a steep drop in temperature, lower exhaust gas temperature, lower exhaust gas pressure, smaller exhaust loss, smaller engine cooling loss and a higher level of efficiency.